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Phase A:

Initial Development Of An Advanced Diagnostic Procedure For Air-Side Retrofits In Commercial Buildings

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ABSTRACT

The objective of this research is to develop a diagnostic approach that involves analyzing monitored whole-building cooling and heating energy use in large commercial buildings in order to determine the effectiveness of air-side energy retrofits. HVAC systems consume energy in excess of the sum total of the building loads. This excess energy use is due to the fact that a single air-handler unit in a HVAC system, having to provide conditioned air at different supply temperatures to different zones in the building, can do so only by resorting to either, (a) a certain amount of mixing of cold and hot air streams as in dual-duct systems, or (b) to reheat at the terminal boxes in single-duct systems. This mixing of cold and hot air streams or terminal reheating results in an energy penalty, which can often be minimized by converting a constant air volume (CAV) system to a variable air volume (VAV) system. Usually, this penalty cannot be entirely eliminated. This report proposes an index, called the Energy Delivery Efficiency (EDE), which characterizes this penalty and rates the energy performance of HVAC systems on an absolute scale. The engineering basis of the EDE approach is developed for both one-zone and two-zone buildings, which allows estimating the variation of the ideal EDE with outdoor temperature for a particular building. Year-long measured whole-building cooling and heating energy use data from two retrofitted buildings are finally used to illustrate differences between actual EDE plots of CAV and VAV systems, and how they compare with the ideal EDE of a two-zone building. The use of this analysis approach to provide diagnostic insights into HVAC air-side energy retrofits in both buildings is also discussed. Logical extensions of this work are finally outlined.

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PREFACE

This report has been prepared for the Existing Buildings Research Program of the Office of Building Energy Research at the U.S. Department of Energy. The purpose of this report is to develop a diagnostic method, together with required simplified calculations, that can be used to determine the effectiveness of air-side retrofits, for example, from constant air volume systems to variable air volume systems. Part A of the project develops and describes the method, the analytical framework for using the method, and presents results of initial application of this method to two large commercial buildings. Part B of the project will further refine the analytical basis of the method (if necessary), test the method with monitored data obtained in 2-4 additional buildings, and will define and test a field diagnostic experimental protocol in order to identify retrofit implementation measures which resulted in the retrofit not performing optimally. Recommendations for future development and use of the method will also be provided.

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DISCLAIMER

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EXECUTIVE SUMMARY

1. Introduction

Of the 82 quads of energy consumed by the US annually, 34 quads are used in existing buildings. Commercial buildings consume about 13 quads annually and up to 50% of this energy use can be avoided by proper conservation. Air-side retrofits are rapidly becoming the major energy conservation feature in programs such as the Texas LoanSTAR program and in other conservation programs nation-wide. Unlike lighting and other energy conservation opportunities (ECOs), there are unresolved problems in air-side retrofits and this is one of the few areas wherein more research is needed. Auditors consistently have difficulty in accurately estimating the energy conservation potential of air-side retrofits in individual buildings. Also, once the retrofit is complete, engineers lack a proper diagnostic approach to evaluate whether the HVAC air-side system has been commissioned and tuned optimally. This project has developed a diagnostic approach based on engineering equations of simplified building and HVAC models and whole-building monitored data which enables both the above issues to be satisfactorily addressed.

2. Theoretical Basis

Comfort energy use requirements in commercial buildings differ significantly from those in residential buildings due to the existence of multiple-zones. While, at any given time, a residence consumes energy either for cooling or for heating, HVAC systems in commercial buildings may simultaneously heat and cool the supply air streams. Thus, the total energy consumed by the HVAC system is higher than the mere sum total of all the individual building loads. This excess energy use can be minimized as in variable air volume systems, but generally cannot be entirely eliminated.

Assume that total thermal cooling energy use (E_c) and total thermal heating energy use (E_h) of the building are measured. Then the value (E_c - E_h) can be viewed as the amount of comfort energy which would have been required had no mixing of cold and hot air streams taken place. This is equal to the building and ventilation loads and is thus a sort of thermodynamic minimum. In reality, the building consumes (E_c + E_h) amount of energy. The Energy Delivery Efficiency index (EDE) resulting from simultaneous heating and cooling is defined as:

$$\text{EDE} = \frac{\text{Thermodynamic Minimum Energy}}{\text{Actual Heating and Cooling Energy}} = \frac{|E_c - E_h|}{E_c + E_h}$$

A value of one implies no mixing while a value of say, 0.2 implies that five times more energy is currently consumed than required in an ideal building. The EDE approach can be used to rate HVAC performance on an absolute basis similar to the Carnot Efficiency for heat engines.

If daily time scales are assumed for the analysis, the energy flows can be treated as steady-state. Then daily values of EDE as a function of the daily outdoor temperature (the most influential of all variables that impact energy use) can be easily generated from the monitored data. In a one-zone building, air stream mixing NEED not occur and $EDE_{Ideal,1\text{-}zone} = 1$. This ideal is, however, an unrealistic base with which to compare measured EDE values. An adequate treatment would be to assume the multi-zone commercial building to consist of two zones: an exterior zone and an interior zone. An analytical formulation of $EDE_{Ideal,2\text{-}zone}$ has been derived from basic engineering equations governing building heat flows. This model requires but a few macroscopic building and operating parameters, many of which can be deduced from the monitored data itself, while others can be estimated quite accurately based on engineering judgment.

3. Data Analysis

Analyzing monitored data against this ideal index allows the analyst to decide whether the HVAC system is performing adequately or whether fine-tuning or drastic improvements are warranted. The approach is illustrated with monitored data from two commercial buildings having a year's worth of data both prior to and after HVAC retrofit (from CAV and VAV). These two buildings exhibit a number of differences in terms of size, operation and HVAC performance and in this regard are representative of the range to be found in commercial building stock. Figure A includes four frames illustrating how year-long daily measured EDE ratios varied with daily outdoor temperature for the two buildings before and after HVAC air-side retrofits were implemented. We note from Fig. A that the HVAC system in Bldg.1 after the retrofit is operating in a close-to-ideal manner since the data points scatter around and along the ideal line. Note that some of the scatter is due to day-to-day variations in internal loads which the model fails to capture properly since a mean internal load value is assumed for generating the ideal curve. This is not the case for Bldg.2 . Though there has been an improvement in EDE for Bldg.2 due to the retrofit, further energy savings may be possible. A closer investigation in this particular building revealed that the air-conditioning was provided by two dual-duct HVAC systems and one single-duct HVAC system, all of which were originally operated as CAV units. The retrofit consisted of converting only the two dual-duct systems into VAV operation. Thus, the reason why the post-retrofit EDE is low seems to be due to the thermal mixing still occurring in the single duct CAV unit.

4. Extensions

Part A of the project described the method, defined the field diagnostic measurements and analytical framework for using the method, and presented initial results of applying the method to two buildings. Part B of the project will further refine the analytical basis of the method (if necessary), test the method with results obtained in 2-4 additional buildings, and will propose and test a field diagnostic measurement protocol which can identify factors which prevented the retrofit performing optimally. Recommendations for future development and use of the method will also be stated.

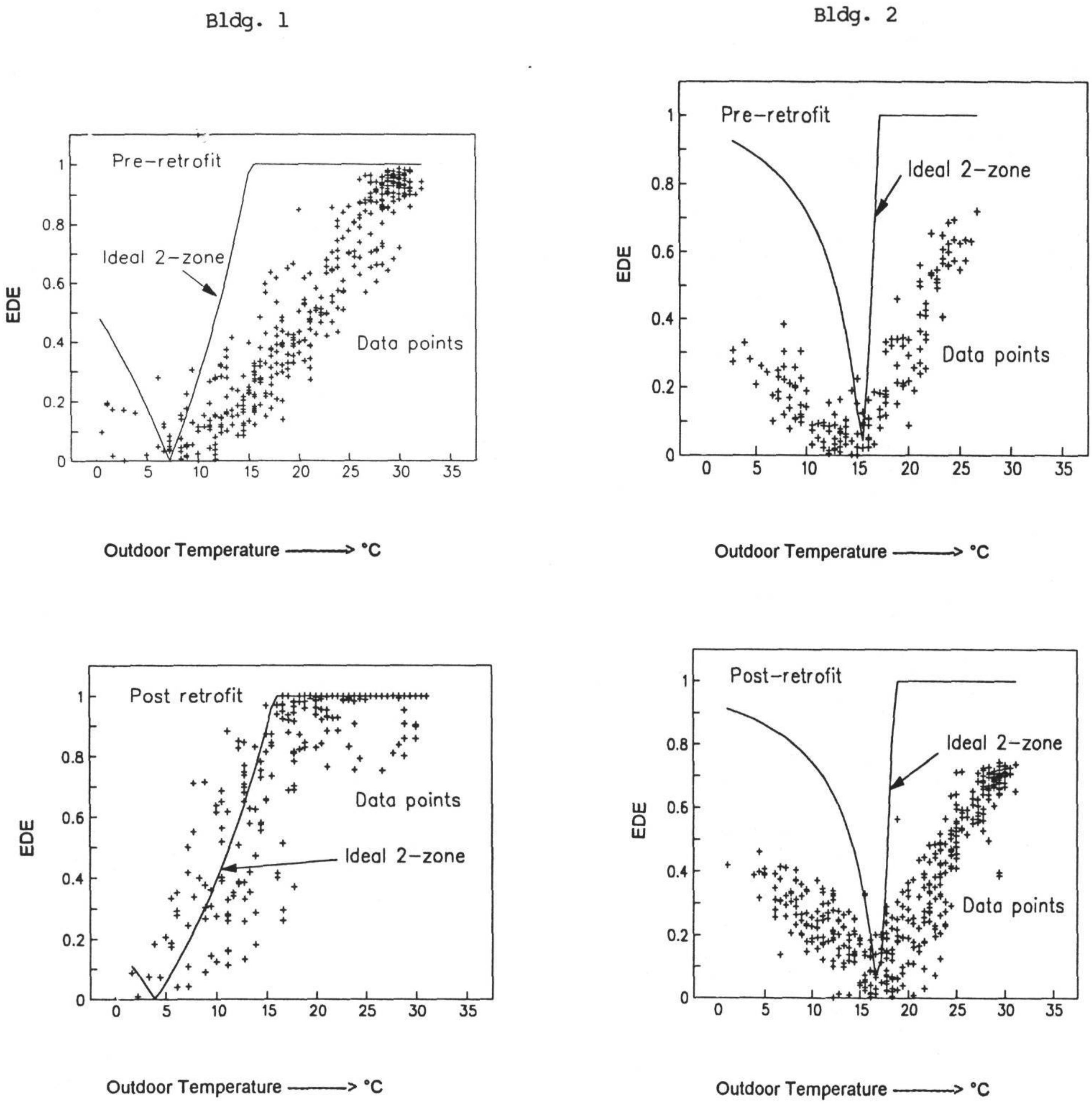

Fig.A Comparison of year-long measured EDE ratios for two LoanSTAR buildings versus $EDE_{Ideal,2\text{-}zone}$ during both pre-retrofit period (with dual-duct constant air volume systems) and post-retrofit period (with dual-duct variable air volume systems). Notice that the retrofit seems to perform satisfactorily in Bldg.1. while further energy savings are possible in Bldg.2.

1.0 INTRODUCTION

The buildings sector is the single largest consumer of energy (about 40%) in the U.S. economy. More efficient use of energy in homes and offices by way of new technologies and better management of lighting, heating and ventilation systems has resulted in annual savings of $45 billion from the US. energy bill (Rosenfeld and Hafemeister, 1988). The commercial building sector is a prime target for energy efficiency improvements as the energy savings potential in this sector has been estimated at about 40-50% (MacDonald et al., 1988). This is in large part due to the fact that (i) by the mid-eighties about two-thirds of the commercial floor space anticipated for the year 2000 has already been in place, and (ii) the energy systems in these buildings (namely lighting and HVAC systems) have been designed and installed at the time when energy conservation was not a prime motivating concern.

In addition to modifications to the building shell and changes in building operation and schedule, retrofits to HVAC equipment and replacement of equipment and lights are widely used energy conservation measures in commercial buildings. Though these are necessary steps, they do not by themselves assure that the building is being efficiently operated on a continuous basis. For this purpose, expert systems (Haberl and Claridge, 1987) have been developed and have reached a stage where several manufacturers offer similar systems on a commercial basis. Although such systems are justified for large buildings, their costs are prohibitive for small- and medium-sized buildings. Consequently, a cheaper alternative that has been proposed is the continuous monitoring of a small number of whole-building energy use channels and analysis thereof in order to identify both retrofit savings and improper building operation (see for example, Haberl and Vajda, 1988). This approach has shown great benefit even for very large buildings since it has proven to be a useful complement to the current generation of EMCS systems. Several energy conservation programs relying on the approach of continuous measurement of a relatively few whole-building energy use channels are underway (for example, the LoanSTAR program, see Claridge et al., 1991). Analysis techniques of the metered data during pre- and post-retrofit periods are described by several researchers, for example, MacDonald and Wasserman (1988) and Kissock et al. (1992).

The purpose of this report is to further advance the techniques by which monitored whole-building energy data in commercial buildings are currently being analyzed. This is done by suggesting a new index for analyzing whole-building energy use which is capable of providing certain powerful and fundamental insights into how efficiently the HVAC system delivers energy to the building zones.

Specific objectives of this study are as follows:

(i) to propose an index, called EDE (Energy Delivery Efficiency), by which the total energy performance of HVAC systems in commercial buildings can be rated on an <u>absolute</u> scale. The approach is akin to the concept of Carnot Efficiency as a way of defining the theoretical limit of heat engines as well as rating the relative performance of different engines. The index would serve as a means of evaluating different generic HVAC system types (for example, Constant Volume Dual Duct or Variable Air Volume systems) as well as permitting the energy performance of a particular HVAC system in a specific building to be assessed against an absolute standard; and

(ii) to illustrate the applicability and the diagnostic ability of EDE with monitored daily whole-building cooling and heating energy use in two different commercial buildings.

2.0 THEORETICAL BASIS

A fundamental difference in comfort energy use in commercial buildings as compared with residential buildings is that, while a residence is seldom cooled and heated at the same time, many HVAC systems in commercial buildings simultaneously heat and cool the supply air. Thus, the total energy consumed by the HVAC system is usually higher than the net or algebraic sum of the individual zonal loads. This excess energy use is caused in two ways:

(a) the presence of distinct zones in the building with different heating and cooling loads. For example, the high internal loads coupled with the absence of skin losses in the interior zones of the building require that cooling be provided to this space even when the outdoor temperature is low enough to require heating in the perimeter zones. Thus, conditioned air must be provided at different supply temperatures to these different zones in the building. Often, the same airhandler must simultaneously meet these requirements. In a dual-duct constant air volume system (CAV), for example, the separate hot and cold air streams are individually heated and cooled first, and then mixed at the mixing boxes of each zone depending on the required load of that particular zone. The net effect of such mixing is that either the cooling coil (or the heating coil) has to meet the NET building load AND the heating (or cooling) load provided by the other coil. This phenomenon is called coil bucking (Stoecker and Jones, 1982). This bucking inefficiency can be minimized, but usually cannot be entirely eliminated even in variable air volume (VAV) systems. Under low building loads, indoor air quality constraints require fresh air intake in VAV systems to be in excess of the minimum modulated flow rate sufficient to meet the loads, and consequently, simultaneous heating and cooling cannot be totally avoided;

(b) due to engineering constraints on how the heating and cooling coils in dual duct systems operate. Temperature sensors placed in the air streams regulate heating and cooling fluid flow such that the exit air stream temperatures are constant. Alternately, for better energy efficiency, the supply temperatures are set according to an outdoor air temperature schedule (Knebel, 1983). These control settings are often not properly tuned to the load variations of the building and energy bucking takes place even in a one-zone building. This penalty can be further reduced by installing a direct digital controller to vary the deck temperatures such that the cold deck temperature is continuously adjusted to meet the cooling needs of the warmest zone and the hot deck temperature is set to meet the heating needs of the coolest zone (Honeywell, 1988).

The ideal HVAC system should only consume the required amount of energy necessary to offset the net building heat gains and losses and to condition the minimum outdoor air intake. This energy requirement is modeled as a one-zone case and is really the IDEAL limit, since it is possible to envision mechanisms (say, a double-bundle condenser heat pump arrangement whereby heat from the interior zone is pumped to the perimeter zone, Stoecker and Jones, 1982) by which buildings with highly variable zonal loads can still be satisfied by HVAC systems designed for operation with one-zone buildings. Though such systems are being installed in mid-sized commercial buildings, these are not currently widespread.

Taking the control volume to include both the HVAC system and the building, and viewing electric loads such as lighting to be generated internal to the control volume, a heat balance yields:

$$Q_b = E_c - E_h \tag{1}$$

where

Q_b - net building thermal load,

E_c - measured whole-building cooling thermal energy supplied by the cooling coil, and

E_h - measured whole-building heating thermal energy supplied by the heating coil.

The value (E_c - E_h) can be viewed as the amount of comfort energy which would be required had <u>NO</u> mixing of cold and hot air streams taken place. This amount is, thus, a sort of absolute thermodynamic minimum. In actuality, the building consumes thermal energy amounting to (E_c + E_h). Consequently, we define a ratio called the Energy Delivery Efficiency (EDE) which rates this amount of simultaneous heating and cooling as:

$$\begin{aligned} \text{EDE} &= \frac{\text{Thermodynamic minimum energy use}}{\text{Actual energy use}} \\ &= (E_c - E_h)/(E_c + E_h) \end{aligned} \tag{2}$$

Alternatively, in order not to have negative values for the efficiency index when the building requires more heating energy than cooling energy, absolute values can be taken as follows:

$$\text{EDE} = |(E_c - E_h)|/(E_c + E_h) \tag{3}$$

The EDE index of an actual system defined by eq.(3) and estimated from measured whole-building cooling energy and heating energy data would lie between 0 and 1, the latter indicating no simultaneous heating and cooling. The building can then be viewed as operated at its thermodynamic efficiency limit. However, EDE does not account for transport losses nor include conversion efficiencies associated with having to convert primary energy into secondary energy.

Kreider and Rabl (1993) have proposed another type of index, the Multizone Efficiency Index (MEI), to account for the inefficiency caused by simultaneous heating and cooling of different zones. Their approach is analogous to ours but differs in that they define <u>two</u> indices: one for heating and one for cooling. For example, the heating MEI is defined as the ratio of the annual heating energy required by the "ideal" one-zone system to the observed annual heating energy. The choice of the annual time scale renders the index location specific and provides less diagnostic insight than that provided by EDE. Also, heating and cooling energy uses are treated as separate quantities in the MEI approach, while they are treated in conjunction in the EDE approach.

Note that the EDE concept could be applied to any time scale: hourly, daily, monthly or even yearly. We shall, however, present the development with <u>daily time scales</u> implicitly in mind in order to be consistent with current analyses of retrofit energy savings (for example, the LoanSTAR program; see Claridge et al., 1991, Kissock et al., 1992) while providing sufficient diagnostic insight. Further, the use of daily data for this analysis has several advantages: (a) it retains the resolution required to observe variation in energy use with climatic conditions and other building parameters; (b) avoids the complexity introduced by thermal mass effects and by the strong diurnal scheduling patterns of internal loads; (c) significantly reduces the amount of data to be manipulated and interpreted as compared to hourly data, while remaining large enough for robust statistical analyses.

The most common type of HVAC retrofit in the LoanSTAR program is from a dual-duct CAV to a dual-duct VAV (Claridge et al., 1991). Thus, most of the discussion in this report would be directed towards these types of HVAC systems. The general conceptual approach can, however, be applied to any type of HVAC system.

3.0 BUILDING LOADS AND IDEALIZED EDE

In order to better appreciate the EDE concept, we need to understand how building loads should vary with climatic and operating conditions. We shall treat two simplified theoretical scenarios:

a) the building is treated as a 1-zone space, and

b) the building is treated as a 2-zone space: one perimeter zone and one core zone.

3.1 One-Zone Model

A heat balance on the building, neglecting mass storage effects, yields an expression for the total heat load Q_b of the building (Knebel, 1983):

Q_b = shell transmission loads + infiltration and ventilation loads (both sensible and latent) + solar loads (both direct and transmission) + internal loads (including gains from people)

By ventilation load, we mean the energy required to raise or lower the amount of fresh air intake to the enthalpy level of the return air stream. Though ventilation load is strictly an HVAC system load and not really a building load, we have chosen to include it with the other building loads for purposes of convenience in the theoretical treatment.

We shall make the following assumptions :

(a) the HVAC system is assumed NOT to have an economizer cycle, (this assumption will be relaxed later). Further, ventilation air flow rate is assumed constant over the year at a specified design value;

(b) the thermostat set point temperature T_{set} is fixed at a mean yearly value implying no daily or seasonal thermostat set-backs or set-ups;

(c) infiltration loads are assumed negligible or considered part of the ventilation loads;

(d) solar gains are a linear function of outdoor dry-bulb temperature (Vadon et al., 1991 and Knebel, 1983);

(e) daily internal loads consisting of heat gains from lights and equipment and from occupants are approximately constant over the year, and are uniform throughout the building;

(f) the ventilation air flow rate, i.e., the fresh air intake is the same for CAV and VAV systems. Note that it is the air flow rate to the space which is modulated in a VAV system, not the fresh air or ventilation flow rate;

(g) ducts are perfectly insulated (i.e., no heat losses) and ducts have no air leakage. Alternately, duct losses can be considered to be part of envelope loads, and

(h) effects of latent loads will be neglected in the current model in order to retain intuitive simplicity while explaining the concept.

Assuming the sign convention that space heat gains are positive, we have

$$Q_{b,1\text{-zone}} = UA_s\ (T_o - T_{set}) + m_v cA\ (T_o - T_{set}) + a_{sol} + b_{sol}\ T_o + q_i\ A \tag{4}$$

where

A - conditioned floor area of the building

A_s - the surface area of the building

a_{sol} - intercept term of the linearized solar loads

b_{sol} - slope term of the linearized solar loads

c - specific heat of air at constant pressure

m_v - ventilation air flow rate (i.e., the inflow of outdoor air) per unit conditioned floor area

q_i - total internal loads per unit conditioned floor area

T_o - the outdoor dry-bulb air temperature

U - overall shell heat loss coefficient per unit surface area.

The above equation can also be expressed in terms of the balance point temperature of the zone (Mitchell, 1983). The balance point temperature (T_b) is defined as the outdoor temperature at which neither heating nor cooling is required by the zone. From eq.(4), with $Q_{b,1\text{-zone}} = 0$, we have

$$T_b = \left(\frac{b_{sol}}{UA_s + m_v c\ A} + 1\right)^{-1} T_{set} - \frac{a_{sol} + q_i A}{UA_s + m_v cA + b_{sol}} \tag{5}$$

In order to visualize how $Q_{b,1\text{-zone}}$ varies with outdoor temperature T_o, it will be more convenient to consider eq.(4) and re-write it as a simple linear function of T_o such as

$$Q_{b,1\text{-zone}} = a + b\ T_o \tag{6}$$

where

$$a = a_{sol} + q_i\ A - UA_s\ T_{set} - m_v cA\ T_{set} \quad \text{and} \quad b = (UA_s + m_v cA + b_{sol})$$

Thus it is clear that one would expect $Q_{b,1\text{-zone}}$ to be linear for all values of T_o. If monitored whole-building energy use data are available, one would be more inclined to determine a and b by regression rather than seek to determine their values from physical building description and from known operating conditions. This, however, may not be strictly valid because: (a) monitored whole-building data, even for one-zone buildings, relates to secondary system energy use and not to the building loads themselves. Only when no simultaneous heating and cooling occur are the building loads and the secondary system energy use equal to each other; and (b) the effect of latent loads, neglected in this discussion, could have the effect of introducing non-linearity in the relationship between $Q_{b,1\text{-zone}}$ and T_o. This aspect is discussed in more detail later on in this report.

In an ideal one-zone HVAC system, mixing NEED not take place because at any given time, the zone calls for either heating only or cooling only. Thus,

$$EDE_{Ideal,\ 1\text{-zone}} = 1 \tag{7}$$

Figure 1 depicts how the ventilation air flow rate (m_v), building loads (Q_b) and the ideal EDE vary with outdoor temperature in relation to the set point and the balance point temperature T_b. Because we assumed that fresh air intake is the same in both CAV and VAV systems, these plots apply to both systems. Since no economizer cycle is present (assumption a), m_v is constant throughout at the design value. The building load is the same as the HVAC system load because the building is treated as an ideal one-zone building. The behavior of the building load line a in Fig.1, given by eq.(6), is linear. Recall that eq.(6) is strictly valid for our simplifying assumptions when latent loads are omitted. Load line b, also shown in Fig.1, is a more realistic representation of how actual building loads vary with T_o and exhibits a segmented linear behavior when latent loads start to appear. Load lines a and b are identical for T_o values lower than the change point T_c. The growing difference between lines a and b with increasing T_o indicates the increasing contribution of latent loads to the total loads (ratios which may reach up to 30-40%, ASHRAE, 1985). Finally note that the loads are zero at $T_o = T_b$ as per the definition of balance point temperature.

$EDE_{Ideal,1\text{-zone}}$ is always 1 for all values of T_o (see Fig.1). To illustrate more clearly how the balance point temperature T_b affects heating and cooling energy requirements, we have plotted $EDE_{Ideal,1\text{-zone}}$ as defined by eq.(2). We see that the index is +1 for $T_o > T_b$ and -1 from $T_o < T_b$. The discontinuity at $T_b = T_o$ represents the switch-over from heating to cooling as T_o increases.

Note that EDE is defined as a ratio. Thus a low value of EDE does not necessarily imply large energy wastage in an absolute dollar amount since the corresponding energy amount itself could be low. This aspect is also addressed later in the report.

3.2 Two-zone model

The one-zone model is generally applicable to residences and small buildings. However, as discussed earlier, most commercial buildings have rather distinct interior and exterior zones. In order to render our discussion more realistic, let us extend the treatment adopted above to the more appropriate two-zone building.

In addition to the assumptions made earlier, we further assume that:

(i) the building is regular in geometry (a rectangle can be assumed for further simplification) with one exterior or perimeter zone and one interior or core zone as shown in Figure 2. Most office and commercial buildings can be conceptually broken down this way (Knebel, 1983). Offices are normally designed adjacent to windows and so form a ring around the perimeter about 5-7 m wide. Corridors adjacent to the perimeter zone could be either lumped into the perimeter zone (if office doors are normally left open), or lumped into the core zone;

(j) the two zones have identical set point temperatures.

(k) ventilation and the internal loads are shared between both zones in proportion to the conditioned floor areas, and

(l) solar and transmission loads affect the perimeter zone only. Note that the choice of daily time scales also helps legitimize the 2-zone assumption.

For the sake of intuitive clarity, we shall again neglect the contribution of latent loads, just as we did in the previous section. We shall simply keep in mind that latent loads become increasingly important at higher T_o values. Let A_{int} and A_{ext} be the floor areas of the interior or core and of the exterior or perimeter zones respectively. The total floor area A of the building is (see Fig.2):

$$A = A_{int} + A_{ext} \tag{8}$$

Then the thermal load on the interior zone

$$\begin{aligned} Q_{b,int} &= \text{internal load} + \text{ventilation load} \\ &= [\, q_i A_{int} + A_{int} m_v c \, (T_o - T_{set})] \end{aligned} \tag{9}$$

The thermal load on the exterior zone

$$\begin{aligned} Q_{b,ext} &= \text{internal load} + \text{solar load} + \text{building shell load} + \text{ventilation load} \\ &= [q_i A_{ext} + a'_{sol} + (UA_s + b_{sol} + A_{ext} m_v c\,)(T_o - T_{set})] \end{aligned} \tag{10}$$

where $a'_{sol} = a_{sol} + b_{sol} \cdot T_{set}$

The slope coefficient b_{sol} of the linearized solar function is normally small compared to UA_s (Katipamula and Claridge, 1992). It is more convenient to treat the term ($UA_s + b_{sol}$) as an "effective" building envelope coefficient which includes the linearized solar contribution.

For the two-zone case, there are two balance point temperatures, one for each zone. These temperatures are easily deduced from eqs. (9) and (10) by setting the thermal loads equal to zero and solving for T_o.

Balance point temperature for the interior zone:

$$T_{b,int} = T_{set} - \frac{q_i}{m_v c} \tag{11}$$

Balance point temperature for the exterior zone:

$$T_{b,ext} = T_{set} - \frac{q_i A_{ext} + a'_{sol}}{UA_s + b_{sol} + A_{ext} m_v c} \tag{12}$$

Note that $T_{b,int} < T_{b,ext}$. When $T_o < T_{b,int}$, thermal loads of both zones will be heating loads, and when $T_o > T_{b,ext}$, thermal loads to both zones will be cooling loads. Under these conditions no mixing of hot and cold air streams NEED occur, and $EDE_{Ideal,2\text{-}zone} = 1$. However, when $T_{b,int} < T_o < T_{b,ext}$, the exterior zone would require heating while the interior zone would require cooling. Thus, even under our ideal operating condition a certain amount of simultaneous heating and cooling takes place in the mixing boxes of each zone, and $EDE_{Ideal,2\text{-}zone} < 1$. The idealized minimum system energy use of a two-zone building is deduced by adding the absolute values of the right-hand terms of eqs.(9) and (10). Thus

$$E_{min,2\text{-}zone} = |Q_{b,int}| + |Q_{b,ext}|$$

$$= |A_{int} m_v c\ (T_o - T_{b,int})| + |(UA_s + b_{sol} + A_{ext} m_v c\ )(T_o - T_{b,ext})| \tag{13}$$

Under this ideal scenario, the system loads E_c and E_h equal the zone loads $Q_{b,int}$ and $Q_{b,ext}$ respectively, and the expression for the ideal EDE for a two-zone building becomes:

$$EDE_{Ideal,2\text{-}zone} = \frac{\text{Thermodynamic Minimum Energy Use}}{\text{Idealized Minimum Two - Zone Energy Use}}$$

$$= (E_c - E_h) / E_{\min,2-zone} \tag{14a}$$

$$= (Q_{b,int} + Q_{b,ext}) / (\,|Q_{b,int}| + |Q_{b,ext}|\,) \tag{14b}$$

As previously, negative values for EDE when $E_h > E_c$ can be avoided by taking absolute values of the numerators of the above equations. A perhaps more generalized form of eq.(14b) is to rewrite it as:

$$EDE_{Ideal,2\text{-}zone} = \begin{cases} (1 - Q_{b,ext} / Q_{b,\text{int}}) / (1 + Q_{b,ext} / Q_{b,\text{int}}) & T_{b,int} < T_o < T_{b,ext} \\ 1 & \text{otherwise} \end{cases} \tag{15}$$

where,

$$\frac{Q_{b,ext}}{Q_{b,\mathrm{int}}} = (\frac{A_{ext}}{A_{\mathrm{int}}})(\frac{T_{b,ext} - T_o}{T_o - T_{b,\mathrm{int}}})(1 + \frac{UA_s + b_{sol}}{m_v c\ A_{ext}}) \tag{16}$$

Larger values of (A_{ext} / A_{int}) would suggest a shell dominated building, i.e., one more influenced by T_o than by the internal loads, and vice versa. Figure 3 illustrates how various quantities vary with outdoor temperature in relation to the set point T_{set}, and both the balance point temperatures, $T_{b,int}$ and $T_{b,ext}$. Since no economizer cycle is present, m_v is constant for all T_o values. The discussion thus applies to both CAV and VAV systems since m_v has been assumed the same for both system types. How the loads for the interior and the exterior zones as well as the total building loads vary with T_o should be noted . The loads $Q_{b,int}$ and $Q_{b,ext}$ for the two zones given by eqs.(9) and (10) respectively are linear with T_o throughout, as in the one-zone case discussed earlier (and shown in Fig.1). However, the total building cooling load line (E_c) has one discontinuity or change point behavior at $T_{b,ext}$ while the total building heating load line (E_h) also has one discontinuity, at $T_{b,int}$.

Note that the quantity (E_c - E_h), representing the NET cooling energy, should equal the building loads following the First Law of Thermodynamics. The net cooling energy in two-zone buildings is effectively similar to $Q_{b,1\text{-}zone}$ for a one-zone building. The net cooling is linear for all values of T_o, provided of course that latent loads are overlooked. The effect of the latent loads is to introduce another change point in E_c, different from $T_{b,ext}$ explained above, which should appear when $T_o = T_c$, the cooling coil surface temperature (around 10^o-15^oC). The two change points for E_c are sufficiently close to be often, but not always, indiscernible when monitored data is examined. To avoid confusing the broad issue, the effect of the latent load has been neglected and the figure drawn accordingly.

How $EDE_{Ideal,2\text{-}zone}$ varies with T_o following eq.(15) is conceptually shown in Fig.3 for a specified value of (A_{ext} / A_{int}) and $[(UA_s+b_{sol}) / (m_v c A_{ext})]$. As before, we have chosen to plot EDE values as both positive and negative numbers for better clarity and consistency with the previous plot. Negative values of EDE simply indicate that net heating energy use is greater than net cooling energy use and vice versa. The index is less than 1 when $T_{b,int} < T_o < T_{b,ext}$ and dips to very low values at certain T_o values. The points of intersection of the (E_c - E_h) line with the temperature axis corresponds to the balance point temperature of the building had it consisted of only one zone, i.e. with perfect inter-zonal air mixing.

3.3 Effect of Economizer Cycle

In order to conserve energy, many HVAC systems operate on an economizer cycle, based on either temperature or enthalpy control (Stoecker and Jones, 1982). Figure 4 illustrates how the ventilation or fresh air intake is affected by a temperature controlled economizer cycle in the case of a one-zone building. In order to avoid latent loads appearing on the cooling coil, the outdoor temperature T_e at which the economizer cycle is activated is usually a few degrees below T_{set}. Since latent load effects have been overlooked in this analysis, we shall assume $T_e = T_{set}$. When $T_o < T_b$ or $T_o > T_{set}$, the ventilation air flow rate m_v is equal to the design value, of the order of 10-20% of the total air flow rate in most buildings. When $T^*_b < T_o < T_{set}$, the entire building air flow rate is drawn from outdoors. Under these conditions, the new balance point temperature T^*_b is given by :

$$T^*_b = \left(\frac{b_{sol}}{UA_s + m_a c\ A} + 1\right)^{-1} T_{set} - \frac{a_{sol} + q_i A}{UA_s + m_a c\ A + b_{sol}} \qquad (17)$$

where m_a is the design total air flow rate per unit floor area.

Because $m_a \geqslant m_v$, it follows that $T^*_b \geqslant T_b$. Consequently, the building load line displays a change in slope when $T_o = T_e$ (see Fig. 4) and drops to zero at $T_o = T^*_b$. In the range $T_b < T_o < T^*_b$, a strategy of modulating the fresh air intake can eliminate the need to supply either cooling or heating to the building. The net effect of the economizer cycle is to save energy as represented by the shaded triangle in Fig.4. Because of practical difficulties in damper controls and in sizing ducts to handle 100% fresh air, most economizer systems operate somewhere between the two extremes of no economizer and a perfect economizer cycle.

This section on the effect of economizer cycles has been presented with the objective of essentially pointing out a specific physical feature of EDE. $EDE_{Ideal,1\text{-}zone}$ is either +1 (during cooling only) or -1 (during heating only) depending on the value of T_o. But contrary to the no-economizer cycle scenario shown in Fig.1, EDE is undefined in the range $T_b < T_o < T^*_b$, since neither heating nor cooling energy is required. This is represented in Fig.4 as a break in the $EDE_{Ideal,1\text{-}zone}$ plot. In this temperature range, energy consumption is ideally zero; in practice, however, there may be a small amount of energy use, in which case EDE given by eq.(2) would turn out to be zero. This should not be misconstrued as being a large performance penalty because the associated energy consumption is low.

In the case of a two-zone building, an economizer cycle will affect the variation of both $Q_{b,int}$ and $Q_{b,ext}$ with T_o (see Fig.3) and make them non-linear. The vent cycle balance point temperature T^*_b can be either higher or lower than $T_{b,ext}$. If it is higher, there will be a range, namely $T_{b,ext} < T_o < T^*_b$, where ideally no energy need be consumed. The ideal EDE is undefined in this case, just as in the one-zone case. If, however, $T^*_b < T_{b,ext}$, E_h may actually increase when $T_o < T_{b,ext}$ since surplus heating has to be provided to the exterior zone to meet the additional cooling load due to higher than stipulated minimum ventilation air. Whether this increase in the heating load of the exterior zone is compensated by the decrease in the cooling energy required in the interior zone is specific to the building and the manner in which the economizer cycle is controlled. In any case, $EDE_{Ideal,2\text{-}zone}$ plots versus T_o will be affected.

3.4 Typical plots of $EDE_{Ideal,2\text{-}zone}$

How $EDE_{Ideal,2\text{-}zone}$ varies with T_0 can be deduced from the equations presented in section 3.2, provided realistic operating ranges for the various parameters can be established. As mentioned earlier, the exterior zone can be taken to be about 5-7 m wide, which for typical multi-story buildings having total conditioned floor areas in the range of 5,000 - 30,000 m^2 yield ratios of interior zone area to total building area in the range 0.3-0.8. Lower (A_{int} / A) values indicate that the building is shell or envelope dominated as against internal load dominated. A typical range of m_a, i.e., air flow rate per unit floor area for a CAV system is 3 to 6 x 10^{-3} kg/s-m^2 (about 0.5 - 1 cfm/ft^2) (Katipamula and Claridge, 1993). Return air is usually 70% to 90% or more of the total air flow rate+ (Stoecker and Jones, 1982). Thus a value of m_v which lies in the middle of these ranges would be 0.9 x 10^{-3} kg/s-m^2 (0.15 cfm/ft^2). Electric consumption from lights and equipment is measured in the LoanSTAR buildings and a realistic range is 10 - 40 W/m^2 (Abbas, 1993). Internal heat gains from people (as well as loads due to air handler motor inefficiencies, Knebel, 1983) need also to be included. Since these loads are relatively small (about 10-20% of daily total cooling loads) and the schedule of lights and equipment closely follows that of building occupancy, a constant multiplicative correction of 1.15-1.25 to the measured lights and equipment consumption would be a simple way of determining total internal loads. Internal load densities on a daily average basis are often in the range 10-40 W/m^2. Also, (UA_s/A) values are typically in the range of 1-3 W/^oC m^2. Solar effects in many of the LoanSTAR buildings are small (in the range of 5% of the envelope transmission losses, Katipamula and Claridge, 1993) and we can assume them to be contained in an "effective" (UA_s/A) value. Finally, T_{set} is usually in the comfort range of 21^o-25^oC.

It must be cautioned that representative ranges of the various parameters chosen above correspond to DAILY averages. Instantaneous or hourly values may be higher than daily values and due care needs to be exercised in the engineering choice of the various parameters while analyzing building monitored data.

The range of variation of the interior balance point can be determined from eq.(11). If T_{set} is assumed to be a constant parameter equal to 22^oC, then $T_{b,int}$ is dependent on only two parameters: q_i and m_v. As shown in Fig.5, its range of variation is from -50^oC to 10^oC. For typical values of q_i = 25 W/m^2 and m_v = 0.9 x10^{-3} kg/s m^2 (0.15 cfm/ft^2), $T_{b,int}$ = -10 ^oC. Increasing q_i or decreasing m_v lowers $T_{b,int}$. Outdoor temperatures in many locations in the U.S. never reach such low values and this explains why cooling energy is required in commercial buildings even during winter.

+ Return air percentage can be much lower in certain types of institutional buildings (for example, chemistry buildings) where large amounts of air pollutants generated indoors need to be vented.

From eq.(12), the exterior zone balance point is a function of four variables: q_i, A_{int} / A, UA_s/A and m_v. $T_{b,ext}$ is relatively insensitive to m_v and so a mean value of 0.9×10^{-3} kg/s m^2 (0.15 cfm/ft^2) can be assumed. How $T_{b,ext}$ varies as a function of the other three variables, assuming $T_{set} = 22^oC$, can be seen in Fig.6. We note that $T_{b,ext}$ decreases with increasing q_i, and with decreasing (UA_s/A) and (A_i / A) values. $T_{b,ext}$ can be as low as 10^oC but normally one would expect it to be about 15^o-18^oC.

$EDE_{Ideal,2\text{-}zone}$ is a function of the following parameters: T_o, q_i, A_{int} / A, UA_s/A and m_v. Again, the effect of the last parameter is small and a mean value of 0.9×10^{-3} kg/s m^2 (0.15 cfm/ft^2) can be assumed. How $EDE_{Ideal,2\text{-}zone}$ is affected by the other parameters is depicted in Figs. 7 and 8. Noteworthy trends are that:

(a) increasing internal load density (q_i) for a fixed value of (UA_s / A) results in a translation of EDE plots towards lower T_o values as well as a small change in the slope of the S-curves (Fig.8);

(b) decreasing envelope loss coefficient (UA_s / A) for a fixed value of q_i also has a similar effect (Fig.7);

(c) the effect of decreasing interior zone to total area (A_{int} / A) results in a decrease in the slope of the EDE curves, i.e., leads to higher EDE values. This is logical since the limiting condition of (A_{int} / A) =0 results in a one-zone case where $EDE_{Ideal,1\text{-}zone}$ has a step change behavior as shown in Fig.1;

(d) for a given value of (UA_s / A) , the EDE plots for different (A_{int} / A) values intersect at two different outdoor temperatures. The asymmetric behavior of the various sets of curves for different (A_{int} / A) values close to $T_{b,ext}$ and $T_{b,int}$ should also be noted:

4.0 APPLICATION TO MEASURED WHOLE-BUILDING DATA

4.1 Description of buildings

To date (as of Dec. 1993), there have been over three dozen buildings in the Texas LoanSTAR program (Claridge et al., 1991) in which retrofits have been completed. Of these, two institutional buildings have been selected (a) for illustrating how the concepts and formulae derived earlier can be applied to real buildings, and (b) to discuss how the EDE approach could be used as a diagnostic tool to evaluate HVAC system performance. The physical characteristics of the two buildings, coded 1 and 2, are given in Table 1. The selection of these two buildings was also based on their different physical characteristics, which were representative of the range of conditions found in our data base. From Table 1, we see that Bldg. 1 has a conditioned floor area of 24,000 m^2 while Bldg. 2 has only 9600 m^2. The glazing fractions and the outdoor air intake fractions are also different in these buildings. Finally, one year's worth of clean daily pre-retrofit and post-retrofit data has been selected for Bldg. 1, while Bldg. 2 has only 8 months of clean pre-retrofit and 11 months of post-retrofit data available. Note that the HVAC retrofits in both buildings consisted of a change-out from a dual-duct CAV system to a dual-duct VAV system. Both buildings are institutional in that they are provided with electricity, chilled water, and steam/hot water from campus utility plants that are separate from the buildings. The chilled water and hot water energy data (in thermal units) and electricity use by lights and receptacles monitored on an hourly basis are summed to daily values and converted into area-normalized units of W/m^2 in all subsequent discussions. A final point of note is that neither of these buildings have economizer cycles and so fresh air intake fractions can be assumed to be almost constant throughout the year (although this fraction may have changed at the time of the retrofits). The equations derived in section 3.2 are hence directly applicable to both buildings.

4.2 Inferring basic parameters.

The various building characteristics and operating parameters necessary to generate $EDE_{Ideal,\ 2\text{-zone}}$ need to be determined either from direct monitoring or by engineering judgment. Let's first try to determine the overall heat loss coefficient of the exterior zone, i.e. the term $(UA_s + b_{sol} + A_{ext} m_v c)$ in eq. (10). The solar effects in both these buildings are small and we shall assume that a'_{sol} is inherent in q_i and that b_{sol} is lumped into UA_s. As mentioned earlier, the overall heat loss coefficient can be determined by regressing the net cooling load $(E_c\text{-}E_h)$ against T_o. Thus, from eqs.(9) and (10),

$$\begin{aligned} E_c - E_h &= Q_{b,int} + Q_{b,ext} \\ &= q_i\, A + a'_{sol} + (UA_s + b_{sol} + A\, m_v c)\,(T_o - T_{set}) \end{aligned} \tag{18}$$

$$= a'' + (UA_s + b_{sol} + A\ m_v c)\ T_o \quad (19)$$

where $\quad a'' \quad = q_i A + a_{sol} - (UA_s + A\ m_v c)\ T_{set}$

Because q_i, UA_s, m_V and T_{set} are assumed constant, $(E_c - E_h)$ should be linear in T_o. This statement is only valid if latent load effects are neglected, as was assumed when eqs. (9) and (10) were derived. The effect of latent loads is to introduce a change point behavior in the net cooling line at T_c as shown in Fig. 1. The slope of the lower portion of the segmented line would more accurately represent the overall heat loss coefficient. Consequently, a more accurate procedure to determine this loss coefficient is to assume a functional form given by a four-parameter (4-P) change point model (Ruch and Claridge, 1992) as:

$$E_c - E_h = \alpha_1 + \alpha_2 (T_o - \alpha_4)^- + \alpha_3 (T_o - \alpha_4)^+ \quad (20)$$

where the four parameters to be determined by regression are $\alpha_1, \alpha_2, \alpha_3$ and α_4, and the – and + signs above the parentheses signify that the particular terms are to be included in the regression only if they are negative and positive respectively. It is easy to infer from eq. (20) that the α_2 denotes the overall envelope heat loss coefficient.

The scatter plots of daily net cooling load versus T_o for both buildings are shown in Fig. 9. Both pre-retrofit and post-retrofit data points are shown as well as the regression lines following eq. (20). In both buildings there is a decrease in the slope of the left-hand segment from pre to post-retrofit periods, due to a decrease in the fresh air intake (i.e., m_v) at the time of retrofit. Results of fitting a linear model (eq. 19) and a 4-P change-point model (eq. 20) to the data sets are given in Table 2. Because the net cooling load has been normalized by the conditioned floor area of the building, the left-hand slopes yield $\left(\frac{UA_s}{A} + \dot{m}_v c\right)$ values. The model goodness-of-fit (R^2 values) are very high and the Root Mean Square Errors (RMSE) are low (see Table 2), indicating excellent fits. An important issue which we stress here is that though R^2 and RMSE improvements from a linear model to a 4-P model are modest, the physical parameter, namely the $\left(\frac{UA_s}{A} + \dot{m}_v c\right)$ values determined by both methods are different. The physical parameters determined from a 4-P regression are much more physically consistent, i.e. closer to values calculated from a constructional description of the building walls, roof and glazing.

The other physical parameters needed for EDE $_{\text{Ideal, 2-zone}}$ are given in Table 3. Internal electric loads are measured, and correction factors of 1.25 and 1.15 to account for occupant and other unmonitored loads (like solar, etc.) have been estimated for Bldg. 1 and Bldg. 2 respectively from data available in LoanSTAR description note-books (Claridge et al., 1991). We note from Table 3 that q_i for Bldg.1 is 40 W/m^2, at the high end, while that for Bldg.2 is only about 12 W/m^2, representative of the low end. Air flow rates in Bldg. 1 have been measured while those in Bldg. 2 have been estimated from equipment performance data and from discussion with building operators. T_{set} values have been assumed based on engineering judgment and $T_{b,int}$ and $T_{b,ext}$ have been calculated from eqs. (14) and (15) respectively. Note that as discussed in section 3.4, $T_{b,int}$ values are lower than those attained by T_0 during winter and that $T_{b,ext}$ values are in the range 15-20°C.

4.3 Data Analysis

Scatter plots of various area-normalized daily energy use data versus T_0 for Bldg.1 and Bldg.2 are shown in Figs. 10 and 11 respectively. Frames (a) and (b) depict how cooling and heating energy use respectively vary with T_0 during both the pre- and post-retrofit periods. Four-parameter change point regression lines are also shown, not so much for predictive modeling purposes but to indicate central tendency of data variation. We note that both buildings exhibit change-point behavior, though it is more pronounced in some cases than in others. The difference in change points in Bldg.1 from pre- to post-retrofit periods is a result of the reset schedule, which was operating improperly during the pre-retrofit period and which was rectified at the time of the retrofit. The above discussion serves to highlight the fact that though theoretical considerations may predict certain behavior in the variation of E_c or E_h versus T_0, monitored data may not always reveal such behavior distinctly. However, a better understanding of what it ought to be theoretically gives an indication of what types of trends and behavior to look for during analysis of monitored data.

Frames (c) in Figs.10 and 11 show how the total energy use (i.e., sum of chilled water and hot water) vary with T_0. Again, the lines drawn through the data points should be taken as trend lines only. We note that important energy savings have resulted in both buildings as testified by the large decrease in the total energy use throughout the year.

Frames (d) in both figures illustrate how the daily EDE ratios, calculated from eq.(2), vary with T_0 and from pre- to post-retrofit period in both buildings. We suspect, based on the discussion in section 3.2, that the retrofit in Bldg.1 has been more effective than that in Bldg.2 though the retrofit in the latter is also saving energy (see frame c). However, an absolute scale of reference to gauge these EDE points is lacking, and is the primary reason for the mathematical treatment of section 3.2 ,and indeed for the rationale of the entire report.

Frames (e) and (f) indicate how the same EDE data points compare with $EDE_{Ideal,2\text{-}zone}$ as determined from eq.(15) using physical parameters shown in Table 3, for pre- and post-retrofit periods separately. Note that absolute values of EDE have been used in both these frames since it has been found that such a representation enables differences in patterns to be gauged more clearly. We notice from Fig.10 that the HVAC system in Bldg.1 is operating in a close to ideal manner since the data points scatter around and along the ideal line. Note that some of the scatter is due to day-to-day variations in internal loads which the model fails to capture properly since a mean internal load value is assumed for generating the ideal curve. This is not the case for Bldg.2 as is clear from frame (f) of Fig.11. Though there has been an improvement in EDE for Bldg.2 due to the retrofit, further energy savings may be possible. Frames (e) and (f) serve well to illustrate the diagnostic superiority of adopting the EDE approach instead of merely looking at data such as those shown in frames (a)-(c).

A closer investigation of Building 2 revealed that the air-conditioning was provided by two dual-duct HVAC systems and one single-duct HVAC system, all of which were originally operated as CAV units. The retrofit consisted of converting only the two dual-duct systems into VAV operation. Thus, the reason why the post-retrofit EDE is low seems to be due to the thermal mixing still occurring in the single duct CAV unit.

It must be made clear that the objective of analyzing monitored whole-building energy use data in the EDE framework is to give the analyst an absolute basis of evaluating, (a) how a specific HVAC system is currently operating, and (b) whether a particular retrofit is performing at or close to the "ideal" level. This approach requires a careful estimation, or inference from monitored data itself, of the various parameters which determine $EDE_{Ideal,2\text{-}zone}$. Alternate data analysis methods that are simple and yet yield robust estimates of these various parameters are issues which need further research and refinement.

5.0 SUMMARY

The basic scientific and engineering principles which dictate energy use in HVAC systems are well known to the professional community through text-books (Stoecker and Jones, 1982), reports (Knebel, 1983) and ASHRAE literature. However, these publications generally emphasize the design of HVAC systems; the analysis and use of monitored whole-building energy data from large commercial buildings, the diagnosis of design or operational errors and ways to increase efficiency have received little attention. The objective of this report is to introduce a new method for analyzing whole-building energy use data which is capable of providing powerful diagnostic insights into the efficiency of the HVAC system in a specific building.

Commercial buildings generally operate with distinct zones, and this causes the HVAC system to consume energy in excess of the optimal amount, namely the algebraic sum of the individual zonal loads. This excess energy use is a penalty which needs to be minimized; however, it can seldom be eliminated. The penalty is dependent on building operating characteristics and climatic conditions. Outdoor temperature is the predominant climatic parameter for both residences (Fels, 1986) and commercial buildings (Knebel, 1983). The approach presented in this report explicitly recognizes these facts, and proposes a mathematical formulation based on a simplified (but realistic) two-zone treatment of building loads. An energy delivery efficiency index (EDE) is proposed by which the energy performance of HVAC systems can be rated on an absolute scale, akin to Carnot Efficiency for heat engines. Analyzing monitored data against this ideal efficiency allows the analyst to decide whether the HVAC system is performing adequately or whether fine-tuning or drastic improvements are warranted. The approach is illustrated with monitored data from two commercial buildings having a year's worth of data both prior to and after HVAC retrofit (from CAV and VAV). These two buildings exhibit major differences in terms of size, operation and HVAC performance, and in this regard are representative of a significant range of the conditions to be found in commercial building stock. Though retrofits in both buildings saved energy, the diagnostic ability offered by EDE indicated that one building (Bldg.1) was performing satisfactorily, while major energy savings were still possible in the other.

The EDE index, just as the MEI (Kreider and Rabl, 1993), could be a useful design tool if indices are considered on a seasonal or yearly basis. The intent of this research was to develop the EDE approach as a diagnostic tool for which case daily time scales are most appropriate. Note also that EDE does not consider conversion efficiencies of primary energy devices, i.e., it is limited in its current development to air-side system performance only. Also, transport losses such as pressure drops in ducts and terminal boxes are not included.

6.0 EXTENSIONS OF CURRENT WORK

Phase B of the project would involve the following issues:

(a) The current theoretical development of the EDE approach deals with sensible building loads only. The importance of latent loads needs to be ascertained and the analytical treatment of EDE should be extended , if necessary, to include such effects.

(b) It is perhaps unrealistic to compare actual EDE against EDE_{Ideal} , which is a building load characteristic. There are inherent energy penalties in the way different generic HVAC systems deliver energy to the building zones. Hence even if a VAV retrofit is performing optimally, some difference between the ideal and actual EDE indices may still exist. It would be more realistic to compare measured EDE ratios against theoretical system EDEs and not against Ideal building load EDEs. In order to do this we need to have (i) simplified but accurate models to predict the performance of different HVAC generic types, and (ii) short-term or one-time measurements of different system parameters (like the cold and hot deck temperatures, the air flow rate,...). Data from 4-6 monitored buildings which we propose to study should yield some indication as to whether this sophistication is useful or not.

(c) Study monitored data from 2-4 buildings other than the two already investigated. Repeat the current analysis and identify those retrofits that are performing to satisfaction and those that are not.

(d) Develop an experimental plan to perform short-term diagnostic tests in those buildings where the EDE analysis suggests possible retrofit improvements in order to detect causes of improper operation.

(e) Perform tests in these buildings and refine test plan, if necessary.

(f) Prepare a comprehensive final document that fully describes the analytical basis of the EDE approach, that suggests how monitored whole building data need to be analyzed in the EDE framework so as to ascertain the effectiveness of the air-side retrofit, and that proposes an experimental plan of how to perform diagnostic tests in retrofitted buildings whose air-side retrofits do not seem to be performing satisfactorily.

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Nomenclature

A	Conditioned floor area of building
A_s	Surface area of building
a,b	Coefficients defined in eq.(6)
a_{sol}, b_{sol}	Intercept and slope terms of linearized solar loads
c	Specific heat at constant pressure
E	Whole-building HVAC system energy use or load
m_a	Total supply air flow rate per unit floor area of building
m_v	Ventilation or fresh air flow rate into building per unit floor area of building
Q_b	Building thermal gain
q_i	Internal loads per unit floor area of building
T_o	Outdoor air dry-bulb temperature
T_b	Balance point temperature of zone or building under normal operation
T^*_b	Balance point temperature under economizer operation
T_c	Cold coil surface temperature
T_e	Outdoor dry-bulb temperature at which economizer cycle is activated
T_{set}	Thermostat set point temperature of the zone or building
U	Overall building shell heat loss coefficient per unit surface area

Subscripts

a	air
c	cooling, cooling coil
ext	exterior zone
h	heating, heating coil
int	interior zone
Ideal	Ideal
min	minimum
o	outdoor
sol	solar
v	ventilation

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Table 1. Description of buildings chosen.

Bldg. Code	Use	Area m^2	No. of floors	Wall type	Glazing fraction of wall	Glazing type	Outdoor air fraction	HVAC schedule	Pre retrofit dates	Post retrofit dates
1	classrooms labs offices	24,000	3.5	Insulated cement block	9%	Single pane	10-20%	24 hrs/d	10/89 11/90	1/92 12/92
2	classrooms offices auditorium	9,600	5	concrete with face brick	10%	Single pane	3-5%	24 hrs/d	10/90 5/91	10/91 12/92

Table 2. Building loss coefficient identified from regression and associated model statistics.

Bldg.	Retrofit Period	Linear Model[a]			4-P Change Point Model[b]		
		$\left(\frac{UA_s}{A} + \dot{m}_v c\right)$	R^2	RMSE	$\left(\frac{UA_s}{A} + \dot{m}_v c\right)$	R^2	RMSE
1	Pre	3.64	0.92	8.69	2.91	0.93	8.11
	Post	2.90	0.87	8.05	2.13	0.88	7.78
2	Pre	2.30	0.90	4.51	1.98	0.91	4.16
	Post	1.87	0.91	4.19	1.30	0.93	3.74

a. Following eq. (19)

b. Left hand slope of fit following eq. (20)

Table 3. Measured and inferred building parameters used to generate EDE Ideal, 2-zone.

Bldg.	$\frac{A_s}{A}$ [a]	$\frac{A_{\text{int}}}{A}$ [b]	Retrofit Periods	q_i [c] W/m^2	Fresh Air Intake %	$\dot{m}_v$ $kg/s-m^2$	$\frac{UA_s}{A}$ [d] $W/m^2{}^\circ C$	T_{Set}^a °C	$T_{b,\text{int}}^e$ °C	$T_{b.ext}^f$ °C
1	0.5	0.70	Pre	40.0	20	1.2×10^{-3}	1.7	22	-12.3	14.4
			Post	40.0	10	0.6×10^{-3}	1.5	22	-43.7	14.8
2	1.4	0.30	Pre	12.5	10	0.5×10^{-3}	1.5	22	-4.0	19.25
			Post	11.0	5	0.3×10^{-3}	1.0	22	-13.7	20.6

a - estimated value

b - estimated value assuming a perimeter corridor of 5 m

c - obtained from monitored data of lights and receptacles and adjusted for occupant and solar loads

d - determined by regression from data using eq. (20)

e - calculated from eq. (11)

f - calculated from eq. (12)

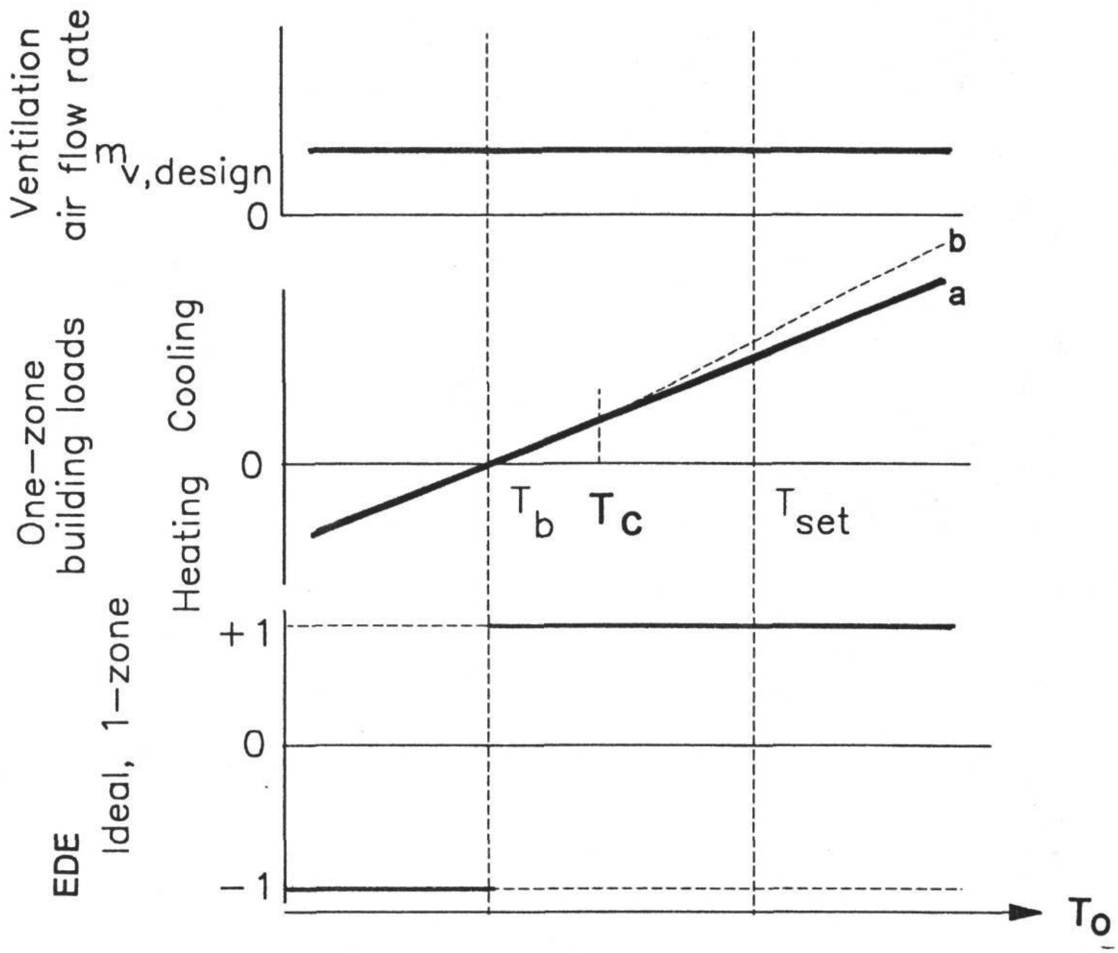

Fig.1 Conceptual variation of ventilation air flow rate, building loads and $EDE_{Ideal,\ 1\text{-}zone}$ with outdoor temperature. No economizer cycle is present and the building is assumed to be a one-zone building. Load line a is linear as given by eq.(6) and neglects latent loads. Load line b is a more realistic representation that includes the effects of the latent loads. The change point T_c shown on the graph is the temperature where line b diverges from line a. The discontinuous variation of $EDE_{Ideal,1\text{-}zone}$ at the balance point temperature is to be noted and represents the point at which the building switches from heating to cooling.

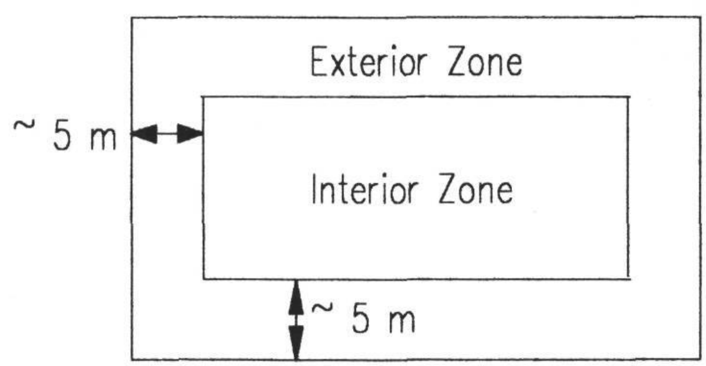

Fig.2 Assumed geometry of a two-zone building with exterior and interior zones.

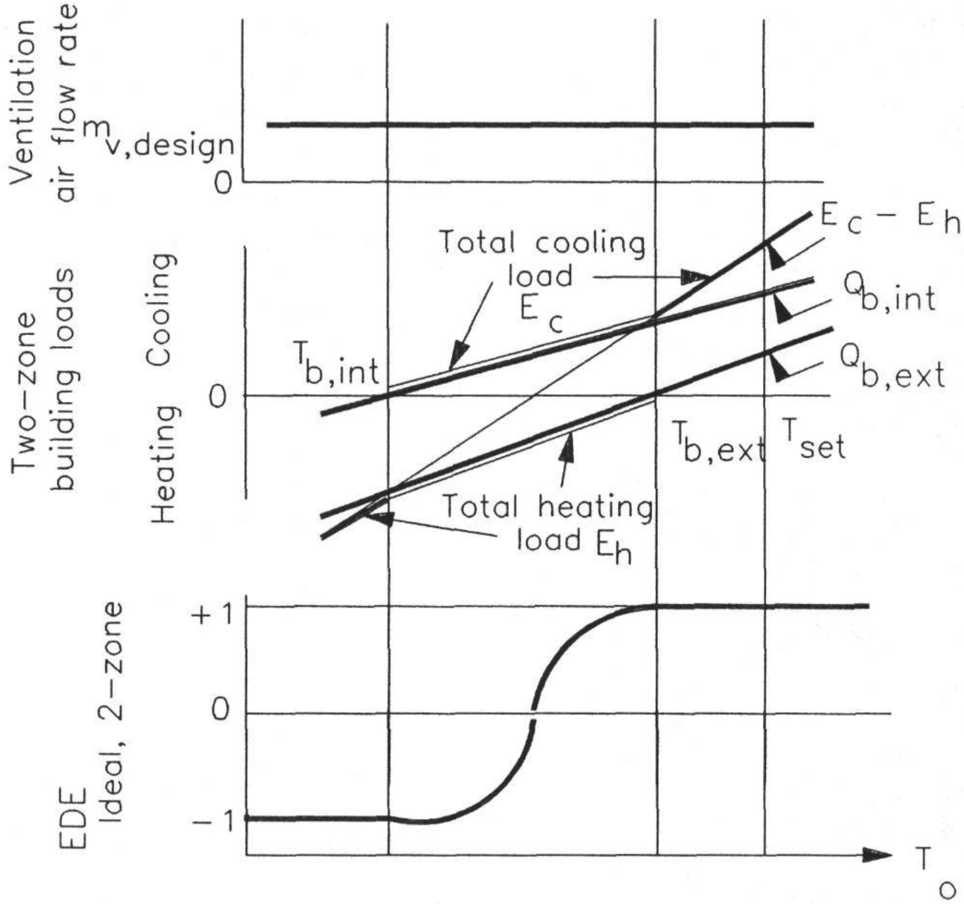

Fig.3 Conceptual variation of ventilation air flow rate, inner and outer zone loads, total cooling and total heating building loads, and $EDE_{Ideal,2\text{-}zone}$ with outdoor temperature. No economizer cycle is present and the building is treated as a two-zone building and latent loads are neglected.

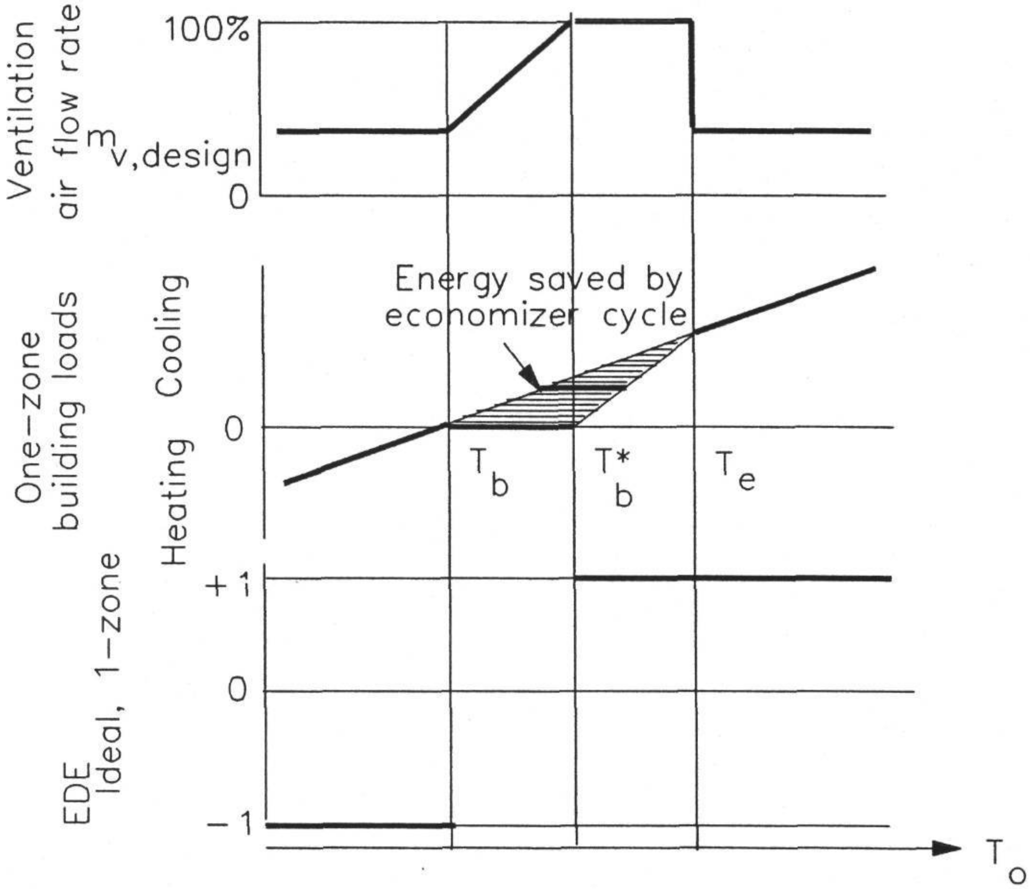

Fig 4 Conceptual variation of ventilation air flow rate, building loads and $EDE_{Ideal.1\text{-}zone}$ with T_o assuming a temperature-controlled economizer cycle. T^*_b is the balance point temperature at which ventilation air is assumed to become 100% of the total building air flow rate.

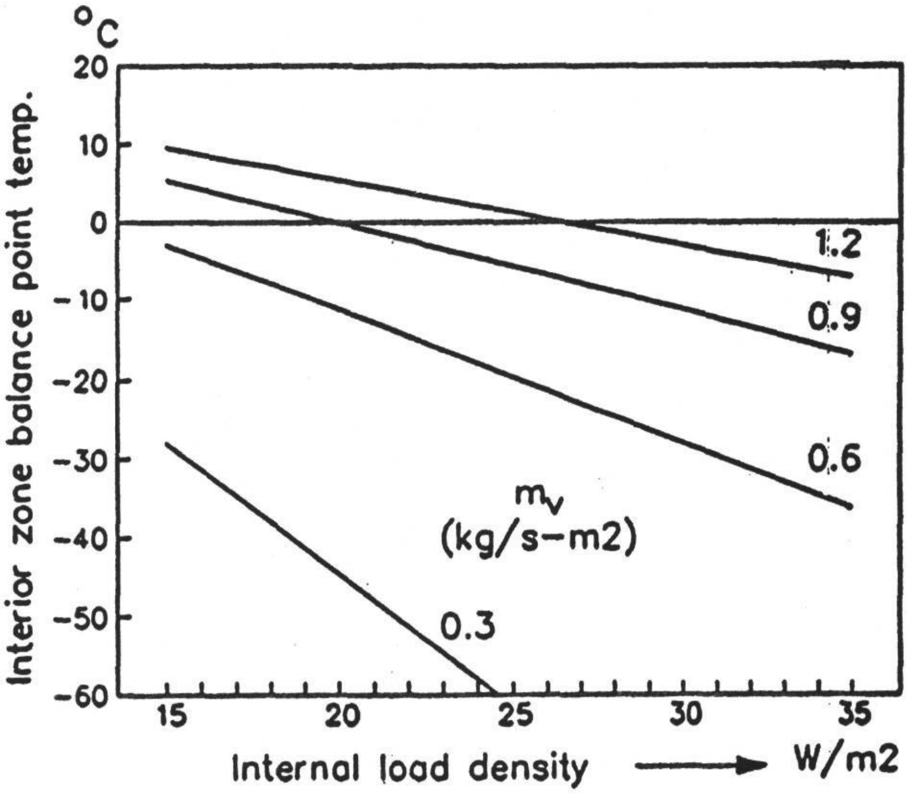

Fig.5 Variation of interior zone balance point temperature $T_{b,int}$ versus interior load density q_i and ventilation air flow rate m_v following eq.(11). A set point temperature of 22°C is assumed. A value of m_v = 0.9 kg/s m^2 corresponds to 0.15 cfm/ft^2.

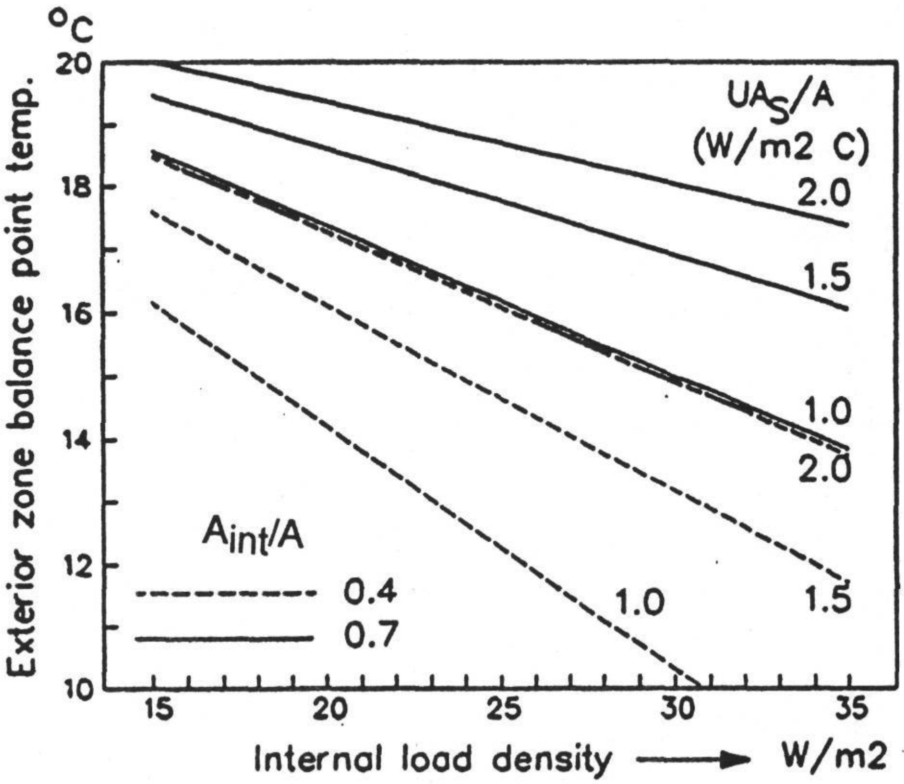

Fig.6 Variation of exterior zone balance point temperature $T_{b,ext}$ versus q_i, (A_{int}/A) and (UA_s/A) following eq.(12). A set point temperature of 22 °C and a ventilation air flow rate m_v of 0.9 x 10^{-3} kg/s m^2 (0.15 cfm/ft^2) are assumed.

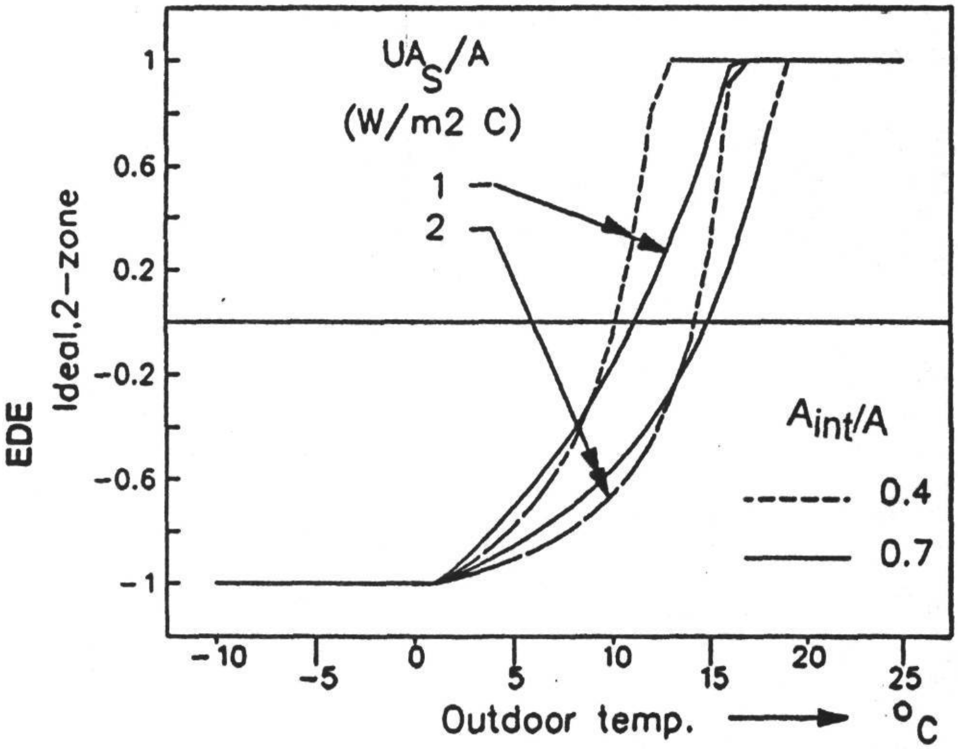

Fig.7 Effect of (A_{int}/A) and (UA_s/A) on $EDE_{Ideal,2\text{-}zone}$ following eq.(15). A set point temperature of 22°C, a ventilation air flow rate of 0.9×10^{-3} kg/s m^2 (0.15 cfm/ft^2), and an internal load density of 25 W/m^2 are assumed.

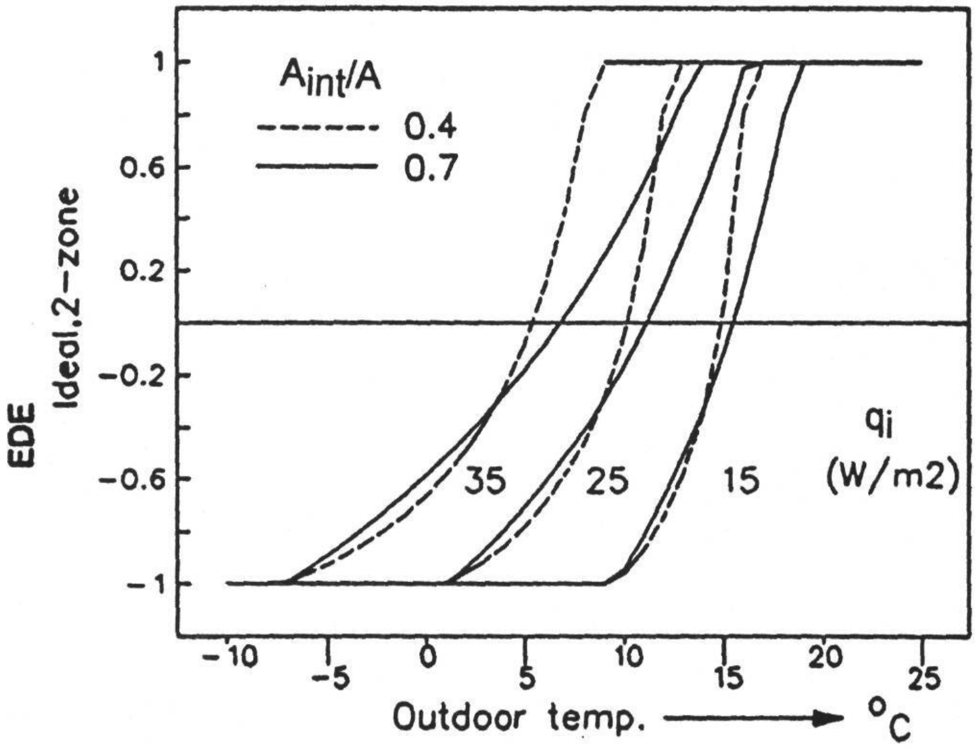

Fig.8 Effect of (A_{int}/A) and q_i on $EDE_{Ideal,2\text{-}zone}$ following eq.(15). A set point temperature of 22°C , a ventilation air flow rate of 0.9×10^{-3} kg/s m^2 (0.15 cfm/ft^2), and an envelope loss coefficient of 1 W/m^2 are assumed.

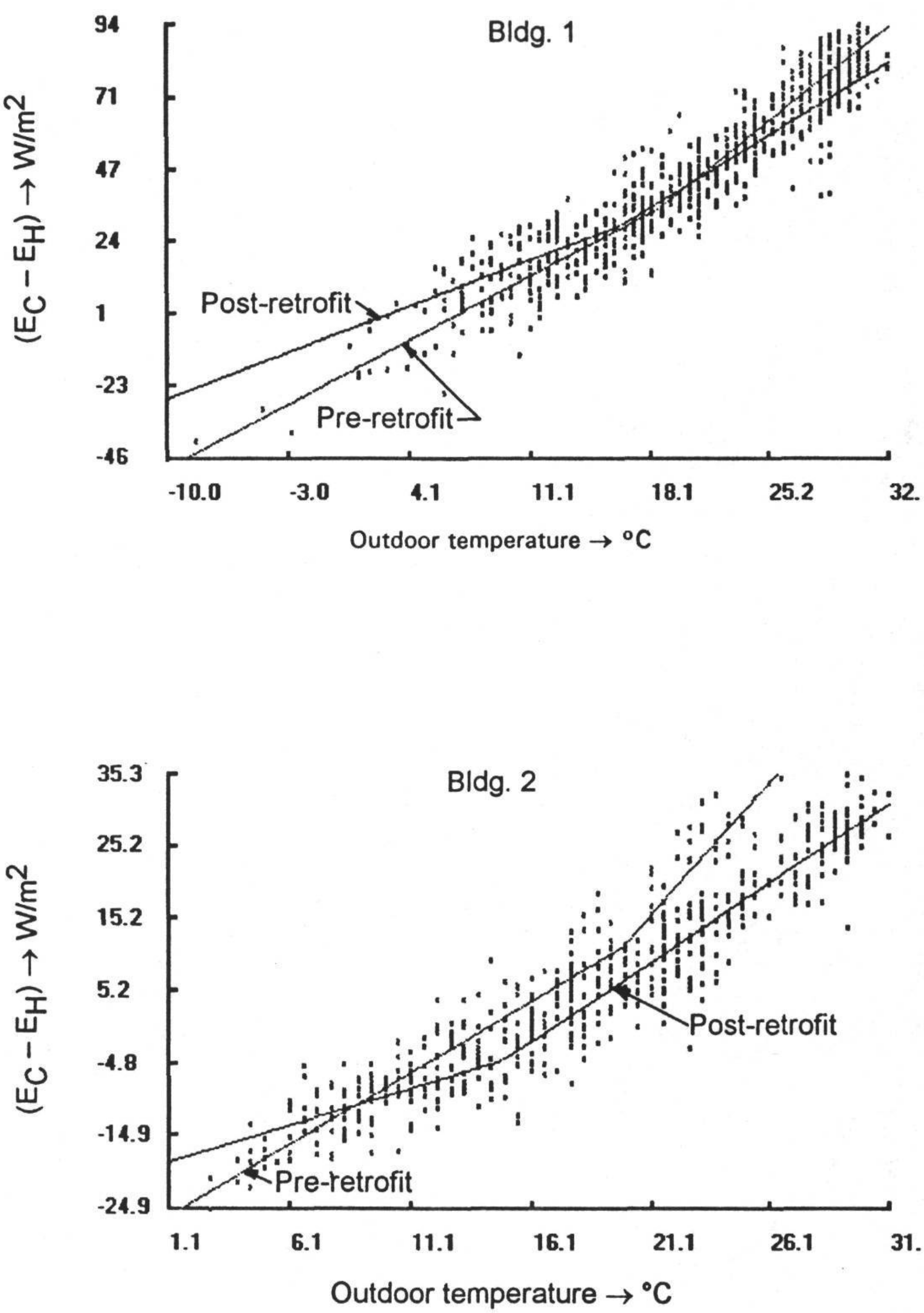

Fig.9 Scatter plots of daily net cooling loads (E_c - E_h) normalized by conditioned floor area versus T_o during pre- and post-retrofit periods. The 4-P change point regression lines (given by eq.20) are also shown.

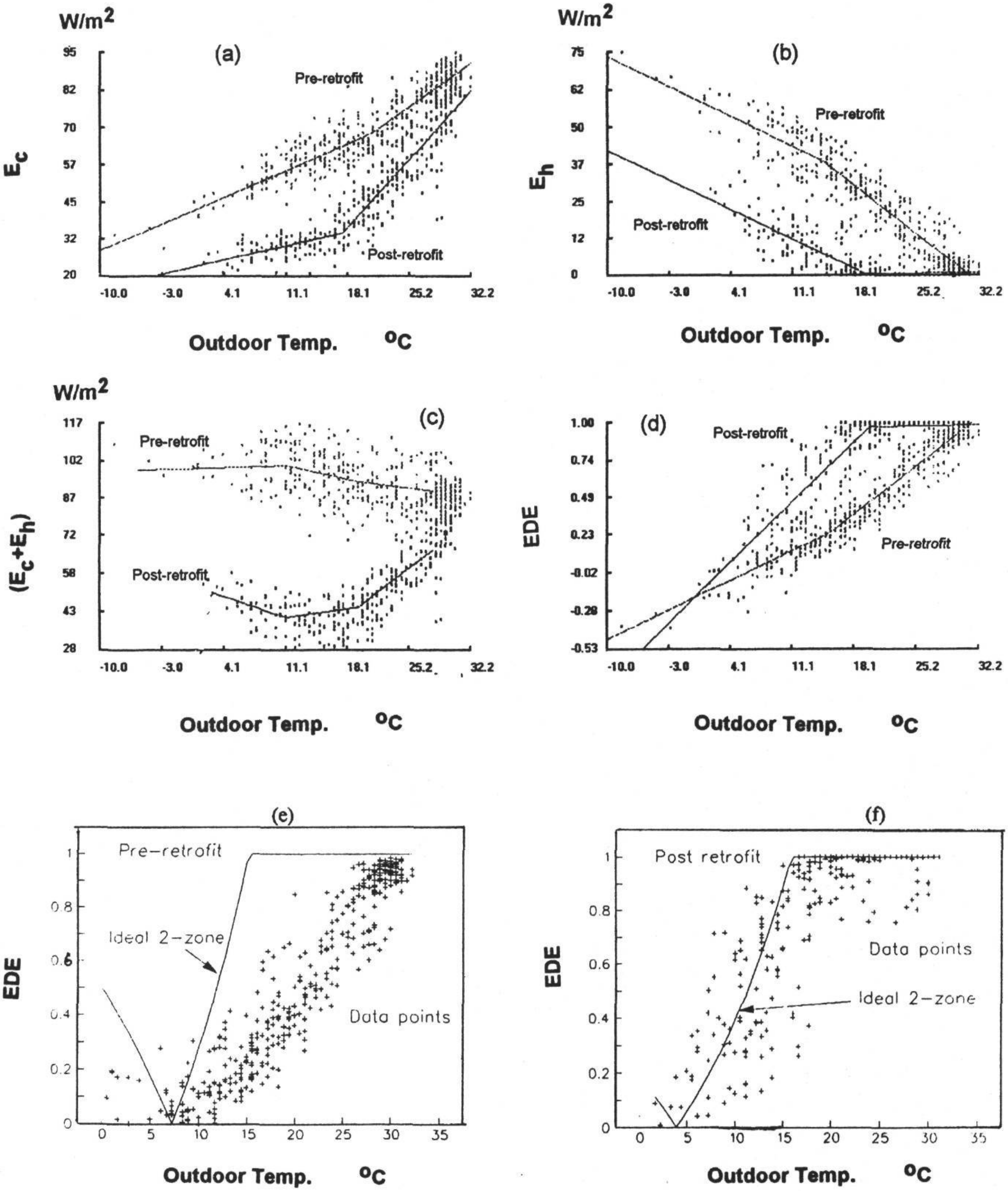

Fig.10. Scatter plots of various area-normalized energy use quantities versus T_0 for Bldg.1. (a) chilled water use, (b) hot water use, (c) chilled water plus hot water use, (d) data-based EDE with 4-P regression lines to show central tendency of data, (e) ideal two-zone EDE plot from eq.(15) versus data points during pre-retrofit period, (f) ideal two-zone EDE plot from eq.(15) versus data points during post-retrofit period.

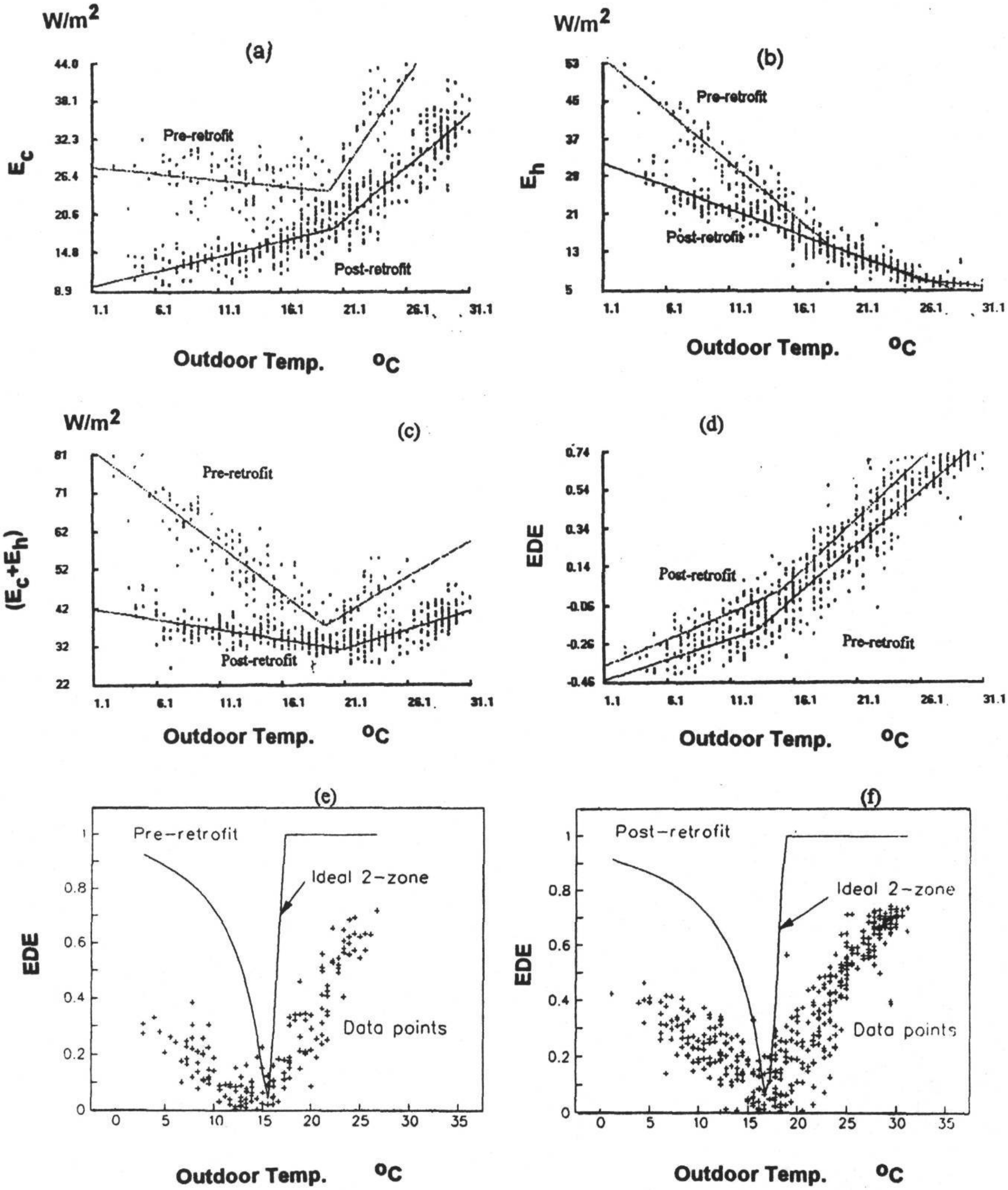

Fig. 11 Same as Fig. 10 but for Bldg. 2.